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An Energy Demand Model for Light-Duty Vehicles, with Concepts for Estimating Fuel Consumption

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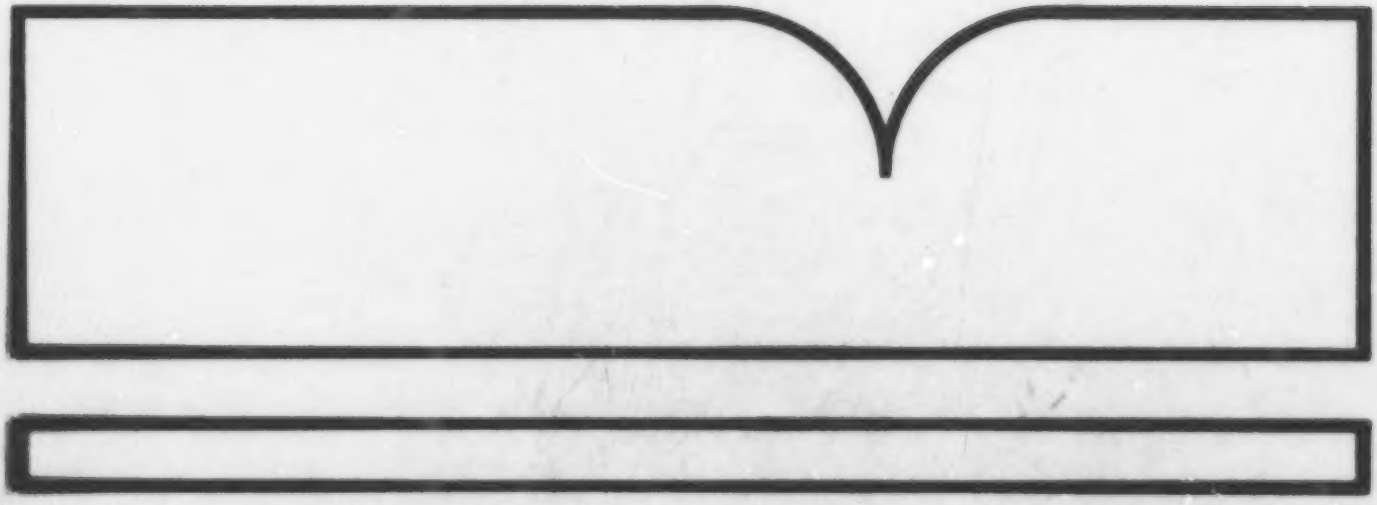

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Technical Report

An Energy Demand Model for Light-Duty Vehicles, with Concepts for Estimating Fuel Consumption

by

Terry Newell

April 1981

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16. ABSTRACT

This report describes the development of the energy demand modeling program, the use of the program, and the verification of using the energy demand concept to estimate vehicle fuel consumption. In addition, several applications of the model are presented and other potential applications are discussed.

17. KEY WORDS AND DOCUMENT ANALYSIS

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I. Introduction

This report presents and discusses a computer program that models the energy demand at the drive wheels of a light-duty vehicle operated over a specified driving cycle. The model is based on direct interpretation of the physical forces that act on a vehicle in motion. In this manner, the model deterministically calculates the energy required to operate the vehicle.

The energy model is also used to estimate vehicle fuel consumption. In this application of the model, the underlying concept is that fuel consumption can be related to energy demand if the characteristics of the drivetrain are known. It is because of the importance of this link that this approach is described as a conceptual means of estimating fuel economy.

There are several advantages inherent in estimating vehicle fuel consumption by this type of approach. The most significant is cost. If a direct empirical approach is taken to observe small changes in parameters affecting vehicle fuel economy, the resulting effects are easily obscured by other factors, and multiple tests are required to isolate and quantify these effects. The "noise" in a computer simulation is limited and predictable; thus a single inexpensive computer modeling provides the desired information.

In his book on computing for scientific and engineering applications, Richard Hamming emphasizes the philosophy that "the purpose of computing is insight, not numbers."[1] The primary intent of this model is to provide the user with added insight into the effects on vehicular energy demand that occur, as vehicle and driving cycle parameters are varied.

The following sections of this report describe the development of the energy demand modeling program, the use of the program, and the verification of using the energy demand concept to estimate vehicle fuel consumption. In addition, several applications of the model are presented and other potential applications are discussed.

II. Development of the Energy Demand Model

A. General

The guiding principle in the development of this model was the efficient determination of energy demand, which then can be used to estimate fuel consumption. Energy demand is computed by maintaining a cumulative total of the vehicle power requirements at each time interval. These powers are computed from knowledge of the forces acting on the vehicle. Therefore, these forces are the fundamental parameters required by the model. This program considers the three primary forces that act on a vehicle that is in motion on a level surface:

$$F = F_T + F_A + F_I \qquad (1)$$

where:

F_T = component of total force due to rolling resistance;

F_A = component of total force due to aerodynamic resistance; and,

F_I = component of total force necessary to overcome inertia.

The rolling resistance component of the total acting force is primarily a function of the tires used on the vehicle. Tire rolling resistance is nearly constant at velocities up to approximately 60 mi/hr.[2] Since the EPA urban (modified LA4) and Highway Fuel Economy Test (HFET) cycles do not specify driving in excess of 60 mi/hr (the top speeds are 56.7 mi/hr and 59.8 mi/hr, respectively), modeling rolling resistance as velocity-independent is reasonable.

Aerodynamic drag increases proportionally with the square of velocity, and the force due to inertia is the product of the mass of the vehicle and its instantaneous rate of acceleration. Thus, the total force acting on the moving vehicle can be expressed[3] by:

$$F = f_0 + f_2 v^2 + m \frac{dv}{dt} \qquad (2)$$

which is a reformulation of equation (1). This is the basic equation of the model. Energy demand can be computed from equation (2) after the force coefficients f_0 and f_2 and the mass of the vehicle, along with a driving schedule, have been supplied.

B. The force coefficients

The values of f_0 and f_2 for use in equation (2) are typically not available in this direct form. Values of these force coefficients can be obtained from three other, more readily available forms of vehicle data. Each of these is particularly well-suited for a given type of simulation by the model, and are briefly outlined below.

1. Road/track coastdown data. If road coastdown data have been collected and analyzed for the vehicles of interest, the resulting coefficients of the acceleration equation can be used to obtain values of f_0 and f_2. This method of determining the force equation coefficients is the best for modeling vehicle energy demand for on-the-road operation.

Analysis of the speed versus time data collected during coastdown testing gives an equation describing the deceleration of the vehicle:[3]

$$A = a_0 + a_2 V^2. \quad (3)$$

Applying Newton's Second Law of Motion and distributing yields:

$$f_0 = ma_0 \quad (4)$$

$$f_2 = ma_2 \quad (5)$$

(Since there is no conventional assignment of units for a_0 and a_2, care in applying unit conversion factors to the right-hand side of equations (4) and (5) is necessary). From these numbers, the energy demand of the vehicle operated on the road according to a given driving cycle can be modeled.

2. Dynamometer coastdown data. If a vehicle-dynamometer coastdown has been conducted for the vehicle in question, the results of that test can be used to determine values for f_0 and f_2. This method of obtaining values for the force coefficients is the best for use in modeling energy demand of the vehicle operated on the dynamometer. The information required is the 55 to 45 mi/hr coastdown time Δt, and the actual (total) horsepower absorbed by the dynamometer, AHP.

The total force acting on the vehicle at the road-dynamometer match point speed of 50 mi/hr is approximated by:

$$F(50) = m\left(\frac{\Delta v}{\Delta t}\right) \quad (6)$$

Of the total power absorbed by the dynamometer (AHP), the power absorber unit (PAU) accounts for the greatest portion. The power absorbed by the PAU is proportional to the cube of the velocity. Therefore, converting dynamometer AHP from horsepower to watts and factoring out the velocity yields an estimate of the component of total force that is proportional to the square of the velocity:

$$F_A = \frac{745.7 \frac{W}{hp} \times AHP}{50 \frac{mi}{hr} \times \frac{1609.3 \text{ m-hr}}{3600 \text{ mi-s}}} \quad (7)$$

The use of dynamometer AHP in equation (7) results in a slight overestimation of the V^2 force term of equation (2), since it includes some minor effects (e.g. dynamometer bearing friction) that may not be proportional to V^2. The PAU setting can be substituted for the AHP in equation (7) if desired; however, this would tend to underestimate the V^2 component. The AHP value appears to be more readily available, and the error in the V^2 term associated with its use is relatively small.

Equations (6) and (7) can then be used to obtain values of f_0 and f_2:

$$f_0 = F(50) - F_A \tag{8}$$

$$f_2 = \frac{F_A}{V^2},\ V = 50 \text{ mi/hr} = 22.35 \text{ m/s} \tag{9}$$

and vehicle energy demand over the driving cycle, on the dynamometer, can then be modeled.

3. Vehicle design parameters. Modeling energy requirements using force coefficients derived from vehicle design parameters, such as aerodynamic drag coefficient, allows certain analyses to be performed concerning design goals and future possibilities. Using this type of input to the model, the drag coefficient (or mass reduction, etc.) required for a targeted fuel economy increase can be estimated. Conversely, the impact on energy demand and fuel economy of intended changes in design parameters can be approximated. In this case, the force coefficients are:

$$f_0 = (mg)(RRC) \tag{10}$$

where:

m = vehicle mass (kg)

g = acceleration due to gravity (9.81 m/s^2)

RRC = rolling resistance coefficient of tires; and

$$f_2 = \frac{1}{2} \rho\ C_D A \tag{11}$$

where:

ρ = air density (kg/m^3)

C_D = coefficient of aerodynamic drag

A = vehicle frontal area.

Equation (2), which forms the basis of the energy demand calculations, can then be reformulated as:

$$F = (mg)(RRC) + (\frac{1}{2} \rho\ C_D A)V^2 + m\frac{dv}{dt} \tag{12}$$

From equations (10) and (11) it is apparent that the force coefficients can be computed given the mass of the vehicle, the rolling resistance of the tires, and the aerodynamic parameters of the vehicle. Note that the important information for calculation of f_2 is the product of the frontal area and drag coefficient.

Equations (10) and (11) also provide a means of estimating values of the vehicle design parameters from road or dynamometer coastdown data:

$$RRC = f_0/mg \tag{10a}$$

$$C_D A = 2f_2/\rho \tag{11a}$$

The rolling resistance coefficient obtained from (10a) will include wheel bearing and other losses, and as such will be slightly higher than the measured value for the tires only. If the vehicle frontal area is known, then an estimate of C_D can also be derived from equation (11a).

C. The Energy Demand

After all of the information necessary for solving the force equation,

$$F = f_0 + f_2 V^2 + m\frac{dv}{dt} \tag{2}$$

has been assembled, the vehicle energy demand can be modeled. This basic equation of the forces acting on the vehicle is solved at each second for the duration of the specified driving cycle.

The physical interpretation of this solution depends upon the signs of the total force at velocity V, F(V), and the inertial term of the right-hand side, m dv/dt. There are three combinations of the signs of these terms that can occur, each representing a different physical situation.

If the vehicle is accelerating or is maintaining a constant nonzero velocity, then $m\ dv/dt \geq 0$ and $F(V) > 0$. In this case, all of the forces acting on the vehicle result in power being demanded of the engine. The current (instantaneous) power demand is obtained by multiplying total acting force by current velocity: $P = FV$.

When the acceleration is negative (i.e., the vehicle is decelerating), the inertia component of total force is also negative. The situation has a different physical meaning dependent on whether the total acting force is positive or negative.

If the acceleration is negative but the magnitude of the rolling resistance and aerodynamic terms is sufficient to result in a net positive acting force, then power is still being demanded from the engine. Physically the vehicle is operating under powered deceleration: the rate of deceleration is slower than that of a "free" coastdown, and power from the engine is required to maintain the prescribed driving schedule.

During rapid decelerations, especially at low speeds, the magnitude of the negative inertia term can equal or exceed the sum of the (positive) rolling resistance and aerodynamic components, resulting in a zero or negative total force F. The vehicle is either freely coasting or braking, and no energy is being demanded from the engine to keep the vehicle moving in the prescribed fashion.

III. Computer Operation of the Model

This section of the report describes the Fortran computer program used to model LDV energy demand, the required input, and the program output. A listing of the program is provided as Appendix A for reference. The statements having line numbers enclosed in boxes are used only to estimate fuel economy from modeled energy demand (See Section IV), and are deleted if only energy demand is desired as output.

A. Input

Two sets of data are necessary for running the basic energy demand modeling: vehicle parameters and a driving cycle. The driving cycles that have been used in the development and testing of the model are the modified (1371 second) version of the LA-4 cycle, which forms the basis of the EPA urban driving cycle (FTP),[4] and the 765 second highway fuel economy test (HFET) cycle.[5] Any driving cycle that is defined by a time versus velocity history can be used.

The driving cycle should be listed as a time-vs-velocity table, with time intervals of one second and velocity given in miles per hour. Conversion of mi/hr to m/s is built into the program; if the velocities are given in units other than mi/hr, one line of the program must be modified or deleted.

The required vehicle information consists of the coefficients of the force equation (2). These can be supplied to or calculated by the program, as described in the previous section. All of the calculations in the program are performed using SI (metric) units; the appropriate units for each program variable are given as part of the program listing in Appendix A.

One other short data file must be attached to the program, containing the number of cases to be run, the length of the driving cycle in seconds, and an indication of the type of vehicle

data to be used and the type of operation (dynamometer or road) to be simulated. The number of cases refers to the number of distinct sets of vehicle data to be modeled over the same driving schedule, and is limited only by time and cost to the user. The length of the cycle in seconds is used to control the number of loop iterations per vehicle. The EPA and SAE standard driving schedules are all under 2000 sec; if a cycle of longer duration is to be used, the dimensioning of the velocity vector V must be increased. The third entry in this file is "1," "2," or "3," corresponding to the use of dynamometer coastdown data, road coastdown data, or vehicle design parameters respectively. Simulation of dynamometer operation is assumed if dynamometer coastdown data is input, and road operation is simulated if the other two types of input data are used. The distinction between road and dynamometer simulation is detailed in the subsection on calculations, below.

B. Calculations

After f_0 and f_2 have been read into or calculated by the model, there is a loop of instructions that is executed once for each second of the specified driving cycle. One pass through this loop is described here.

The driving cycle has been input to a vector V, where V(i) is the velocity at time t = i, $1 \leq i \leq$ number of seconds in cycle. The mean velocity for the i^{th} second and the acceleration during that second are calculated as:

$$\overline{V}(i) = [V(i) + V(i-1)]/2 \tag{13}$$

$$dv/dt(i) = V(i) - V(i-1) \tag{14}$$

The total force acting on the vehicle during the time segment t = i is then the sum of the rolling resistance, aerodynamic, and inertia components:

$$F(i) = f_0(i) + f_2 [\overline{V}(i)]^2 + m \, dv/dt(i) . \tag{15}$$

Within the model, the primary difference in the simulations of dynamometer and road operation is in the handling of the mass in computing total acting force by equation (2) or (15). If road operation is to be simulated, then the gravitational mass of the vehicle is entered; the equivalent mass simulated by the dynamometer flywheels is entered for dynamometer simulations. There is also a difference between these simulations in the estimation of the effective equivalent masses of the rotating tire-wheel-brake assemblies, which are required for calculation of the inertia component of total acting force.

In either type of simulation, if an experimentally measured or calculated value of the effective equivalent mass of one rotating

tire-wheel-brake assembly is known, an appropriate adjustment can be made to the vehicle mass and entered as part of the vehicle data. When this information is not known, as is generally the case, standard (default) estimates are used.[3] These estimates are built into the program:

$$M = 1.035\ m \qquad \text{(road)} \tag{16}$$

$$M = m_e + 0.018\ m \qquad \text{(dyno)} \tag{17}$$

where:

M = total effective mass of vehicle (road simulation) or of vehicle-dynamometer system (dynamometer simulation)

m = gravitational mass of vehicle

m_e = equivalent mass simulated by the dynamometer flywheels.

The contribution of the rotating wheels to the total effective mass is, of course, less on the dynamometer than on the road, since only the two driving wheels are rotating during dynamometer operation.

Since the acceleration dv/dt is negative if V(i) < V(i-1), the inertia component of total force is frequently negative. After calculating the force acting on the vehicle by equation (15), the model tests the signs of the total force and the inertia component. As was discussed in the previous section, there are three possible outcomes of this dual check: 1) both are nonnegative, 2) the inertia term is negative but the total force remains positive, and 3) both are negative. These three cases are treated seperately in the model, as described below.

B. Three Modes of Operation

In the first case, where both the total force and its inertia component are positive, all of the acting forces result in power being demanded of the engine. This condition is identified in the program as "A"-mode, and can be physically interpreted as acceleration or steady-speed cruise.

A cumulative total of all "instantaneous" power demands represents the total energy demand. When all of the terms of equation (15) are nonnegative, the maintenance of individual sums representing rolling resistance, aerodynamic, and inertia demands corresponds to the actual physical breakdown of energy demand.

The second case, where the inertia component is negative but of insufficient magnitude to provide for all of the demand of the rolling resistance and aerodynamic components, is identified in the program as "B"-mode. Vehicle operation in "B"-mode can be physically interpreted as deceleration without braking. Stored kinetic energy resulting from the motion of the vehicle is used to

overcome part of the acting rolling and aerodynamic resistances, but there is still a net positive force to be overcome.

When the vehicle is in the "B"-mode of operation, the total net power demand, P = FV, is added to the cumulative total just as is done in "A"-mode. However, the individual contributions to the total energy demand resulting from rolling resistance and aerodynamic losses must be treated differently than when in "A"-mode. The tire and aerodynamic energy demands cannot be simply added to their respective component sums, since this results in the sum of these energy demand components exceeding the total vehicle demand. Consequently, the proportion of the corresponding steady-speed energy demand that is required is calculated, and this proportion of the rolling resistance and aerodynamic terms is added to their respective component sums. That is, the actual total acting force is given by:

$$F[V(i)] = f_0 + f_2 [V(i)]^2 + m\, {}^{dv}/dt(i) \tag{18}$$

where ${}^{dv}/dt(i)$ is negative. During vehicle operation at the corresponding steady speed, the total acting force would be:

$$F'[V(i)] = f_0 + f_2 [V(i)]^2 \tag{19}$$

The proportion of the total steady-speed force that is required at V(i), in "B"-mode, is therefore the ratio of the actual total force to the corresponding steady-speed force:

$$F[V(i)]/F'[V(i)] \tag{20}$$

Thus, it is this proportion of the rolling resistance and aerodynamic losses that are added to the respective component sums:

$$F_T' = F_T[F(V(i))/F'(V(i))] \tag{21}$$

$$F_A' = F_A[F(V(i))/F'(V(i))] \tag{22}$$

In "B"-mode, the inertia term of equation (18) is negative; no energy is being demanded of the engine to overcome inertia. Therefore, the component sum representing energy demand due to inertial effects (kinetic energy) is not incremented.

In the event that the inertia component of force is negative and $-F_I > F_T + F_A$, so that $F < 0$, no power is demanded from the engine. This is indicated in the model as "C"-mode, and is physically interpreted as the vehicle braking. In "C"-mode, none of the component sums representing individual energy demands are incremented.

All of these calculations in the model are performed in SI units; therefore, integration of power demand over time to yield energy demand reduces to simply summing the power demands for each second (without problems with units and conversions). Similarly, summing V(i) in m/s yields the total distance traveled in meters.

It should be noted that it is the existence of the condition identified in the program as "B"-mode that necessitates the second-by-second calculations of the model. If either of these simplifying assumptions are made: a) all of the kinetic energy of vehicle motion is returned for use in overcoming rolling resistance and aerodynamic losses, or b) none of this kinetic energy is returned for useful work; then the equations used to determine energy demand reduce to a set of definite integrals that depend only on the characteristics of the driving cycle. Since the vehicle parameters would have no effect on the solution of these integrals, they can be solved once for any given cycle. The solutions of these integrals and the vehicle parameters could then be combined algebraically to yield the tractive energy requirement of the vehicle over that cycle.

C. Output

The program output can be considered in three parts: the vehicle information, the energy demand and its breakdown, and auxiliary information. These are briefly discussed below.

The vehicle information listed includes the identifying numbers (a case number for that run and an arbitrary vehicle identification number). The input information is listed for reference, and as a check on input accuracy. If vehicle design parameters were supplied, they are listed with the calculated values of the force equation coefficients. If the force equation coefficients were derived from road or dynamometer coastdown data, they are listed along with backcalculated values for the vehicle design parameters from equations (10a) and (11a). The data in this block are labeled to indicate what was supplied as input and what was calculated by the program.

Energy demand has already been discussed; the total and each of the contributing terms are labeled and listed. In addition, the algebraic sum of all of the inertia component energy terms is printed. In any driving cycle with equal initial and terminal velocities (zero, for the EPA cycles), the true algebraic sum of all kinetic energy terms is zero. Therefore, the value printed should be near zero; any significant deviation from zero is indicative of errors in the modeling program.

A few other lines of information are also output. The time (sec) spent in each of the three "modes" of operation discussed earlier is listed. The time spent in "A"-mode is a function of the acceleration characteristics of the driving cycle used, and is vehicle-independent. For the EPA urban and HFET cycles, "A"-mode consumes 544 sec and 319 sec, respectively. The division of the remaining time between the "B" and "C"-modes is dependent on the mass, the tire rolling resistance, and the aerodynamic drag of the specific vehicle.

Distance traveled over the cycle is computed in meters and converted to miles, and both are printed. The maximum power

demanded by the vehicle at the drive wheels is listed, along with the time of its occurrence within the cycle. The maximum power demands for vehicles over the LA4 cycle occur during the 195th or 196th seconds, when accelerating at a rate of approximately 3 mph/sec from velocities of 30-35 mi/hr. Different vehicles experience peak power demand in the HFET cycle at several different times, depending on the relative contributions of the aerodynamic (V^2) and inertial (dv/dt) terms.

D. Basic Error Check

A check for gross error in the computer program consisted of modeling the energy demand of two base-value vehicles and several odd variations of those vehicles. The base vehicles used for this were a 1980 model Volkswagen Rabbit, representing a small car, and a 1981 model Ford F150 4x4 pickup truck, representing a relatively large LDT. The three primary vehicle descriptors (m, RRC, CDA) were then set to zero, singly and in pairs, and the corresponding energy demands modeled. In these configurations, the energy demands of the vehicles can be checked algebraically without using the model.

A vehicle having a mass of zero, while maintaining its associated gravitational weight, would require no kinetic energy over the cycle: the inertia component F_I of total acting force would always be zero. (Maintaining the gravitational weight associated with the original mass is necessary to keep the energy required for overcoming rolling resistance from being affected.) Elimination of the kinetic energy requirement means that the total energy demand becomes the sum of the energy required to overcome the rolling and aerodynamic resistances.

If the base vehicle maintains its mass and weight, but is assumed to have a frontal area or drag coefficient of zero so that $C_D A = 0$, then no energy is required to overcome aerodynamic drag. Similarly, if the rolling resistance of the tires and tire-wheel-brake assemblies is zero while mass, weight, and aerodynamic characteristics are unchanged, then no energy will be required for overcoming rolling resistance.

Setting two of these three primary descriptors equal to zero at a time, three other "dummy" vehicles are generated. Each of these last three vehicles have a total energy demand equal to one of the components: rolling resistance, aerodynamic drag, or kinetic. Running these "dummy" cases through the model revealed no errors in the handling of equation (15) and its terms.

To this point, discussion has focused on the concepts that form the basis for the calculation of energy demand, and on implementation of the model as a computer program. The last sections of the report present several suggested applications of the program.

IV. Uses of the Model

The use of vehicular energy demand modeling can be divided into two general groupings: prediction of the relative energy requirements of vehicles of the future, and as an aid in both the selection of vehicles for test programs and the evaluation of test results.

A. Future Vehicle Performance

As was briefly discussed in Section II, specification of four vehicle design parameters (mass, frontal area, coefficients of rolling resistance and aerodynamic drag) allows calculation of the force equation coefficients, and hence of cycle energy demand. One major advantage of any modeling exercise is the ability to consider systems that may not be available for direct empirical investigation. One of the first uses of this model was to predict the fuel economy that may be possible with current and future optimized technology vehicles.

Assumptions. Several assumptions were made to simplify and enhance prediction of fuel consumption from modeled energy demand of the optimized vehicles. Foremost among these was the use of a continously variable transmission (CVT). Such transmissions have already been successfully installed in some European vehicles, and are currently under further development by Borg-Warner.[6] Furthermore, it was assumed that the CVT would be controlled, probably through the use of microprocessors, to always seek the point of maximum engine efficiency possible under the required loading. A final related assumption was that the transmission and drivetrain system had a constant efficiency of ninety percent; that is, only one-tenth of the engine power available at the flywheel is lost before reaching the axle of the drive wheels.

Under these assumptions the model can be used to directly calculate fuel consumption, if fuel consumption rates are given as a function of demanded power. This information can be obtained from a simple one-dimensional engine map. A set of points representing fuel consumption as a function of power demand are taken from this map, and linear interpolation is used to determine fuel consumption at intermediate levels of power demand.

Optimized Vehicles. Three general size categories of vehicles were chosen for this investigation. In order of increasing size and weight, these were (i) two-passenger vehicles, (ii) four-passenger vehicles, and (iii) five-or-six-passenger vehicles and light-duty trucks, combined in one group. For each of these classifications based on size, two levels of technology, described as current best (CB) and advanced technology (AT) were considered. The technological level refers to values taken for the four design parameters required to calculate the coefficients of the force equation. The CB values have been achieved in vehicles available

today, although not necessarily all simultaneously in a single model. Advanced technology is better, in the sense of lowering energy demand, than technology in use today; however, this can be achieved within the framework of current technology. The values chosen for the design parameters of these vehicles are presented in Table 1.

A few comments regarding the values assumed for the vehicle parameters for the current best (CB) and advanced (AT) levels of technology should be noted. In the case of vehicle mass, there are production vehicles representing each of the three size classes discussed having masses near or below the masses listed for the CB vehicles. In the two-passenger category, the MY1980 Daihatsu Max Cuore, with a mass of 545 kg,[7] is already nearer to the AT than to the CB mass. The four-passenger MY1980 Audi L, with a carrying capacity of about 1,000 lb, has only 910 kg mass.[7] This is almost exactly the mass assumed for the CB four-passenger vehicle. One version of the Volkswagen Jetta, the GLI, can carry nearly as great a load (948 lbs), yet its mass is less than 800 kg in a 2-door version. The mass of the MY1980 Volkswagen front-wheel-drive diesel pickup truck, a vehicle fitting the largest of the three size categories discussed, is only 928 kg.[8]

Rolling resistance of automobile and light truck tires is frequently expressed as a rolling resistance coefficient (RRC). The RRC indicates the units of rolling resistance force (lb, N) per thousand units of vertical load. The assumed current best value, 0.008, is already being attained, at least in some low rolling resistance tires. In an ongoing EPA test program to measure the rolling resistance of tires, data collected to date indicate that there are tires widely available in today's marketplace with flat-surface RRCs in the range of 0.0082 to 0.0090,[9] when tested in accordance with the EPA Recommended Practice for Determination of Tire Rolling Resistance Coefficients (80 percent of design load, at 35 psi).[10] The continuing efforts by the tire industry to reduce rolling resistance seem likely to result in tires having RRCs at or below the 0.007 assumed for advanced technology.

Aerodynamic drag coefficients lower than the assumed current best value of 0.4 are available in some vehicles today, and coefficients lower than the assumed advanced technology value of 0.3 will be available in the relatively near future. Recent research conducted by Volkswagenwerk AG indicates that reductions in drag coefficients of as much as 45 percent over today's range (about 0.35 to 0.55) should be achievable with mass-produced passenger cars.[11] Ford Motor Company is continuing development of a prototype vehicle, currently known as Probe, which has a drag coefficient of 0.22.[12] On the basis of these statements, the assumed values of 0.4 (CB) and 0.3 (AT) certainly appear feasible.

The reductions in vehicle frontal area listed in Table 1 are relatively small. Lower vehicle masses and improved aerodynamic

Table 1

Vehicle Size Class	Assumed Level of Technology	Mass* (kg)	RRC	C_D	A_2 (m^2)	C_DA (m^2)	f_0 (N)	f_2 (kg/m)	Engine Maximum Power (hp)	Scaling Factor
Two-passenger	Current best (CB)	680.4	0.008	0.4	1.7	0.68	53.3	0.398	25	0.357
	advanced (AT)	476.3	0.007	0.3	1.6	0.48	32.7	0.281	18	0.257
Four-passenger	CB	907.2	0.008	0.4	1.7	0.68	71.1	0.398	31	0.443
	AT	635.0	0.007	0.3	1.7	0.51	43.6	0.298	23	0.329
5 and 6-passenger and 'personal' light trucks	CB	1134.	0.008	0.4	2.0	0.80	88.9	0.468	38	0.543
	AT	793.8	0.007	0.3	1.9	0.57	54.5	0.334	28	0.400

* Values in this column are test weights. Curb weights would be approximately 136 kg (300 lb) lower.

design are likely to result in the assumed decreases in frontal areas.

Engine Sizing. Maximum power outputs required for the engines of these six vehicles were calculated to allow adequate performance with very little excess power. The minimum performance conditions were (i) that the vehicle be capable of a 55 mi/hr cruise on any roadway having no more than a five percent grade, and (ii) that zero to 50 mi/hr acceleration take no longer than fifteen seconds. The first of these conditions proved to be the more stringent: if the engine is capable of maintaining the vehicle speed at 55 mi/hr on a five percent grade, then zero to 50 mi/hr acceleration will take approximately thirteen seconds.

The power required to meet the most severe of these conditions was then increased to allow for the power demand of accessories that do not contribute to vehicle motion. The increase for accessories was two hp for the two-passenger vehicle, three hp for the four-passenger, and four hp for the 5/6-passenger cars and personal recreational light trucks. These values represent estimates of the maximum accessory power demand, not the actual power being demanded for accessories while driving. In estimating energy demand and fuel economy for these vehicles, it was assumed that an average in-use accessory power demand of 0.5 hp was operating continuously.

The total power requirements were then rounded up to the next integer horsepower levels. The maximum power output of the engines derived from these requirements ranged from 18 hp for the advanced technology two-passenger vehicle to 38 hp for the current best technology 5/6-passenger vehicle. These values are listed in Table 1.

Fuel Consumption. The final step necessary was "development" of fuel consumption versus power output maps for the optimized engines of these vehicles. The logical starting point was a small, fuel-efficient engine for which the required data are currently available. One such engine is the seventy horsepower turbocharged prechamber diesel engine manufactured by Volkswagen.[13] However, the rated power of this engine, 70 hp, is well in excess of the maximum power requirements of all six of the optimized vehicles when driven over the EPA driving cycles. Consequently, the engine maps for these vehicles were obtained by linear scaling of this map. That is, if an engine with a maximum power of 30 hp was required, it was assumed that this engine would develop 30 hp with the same thermal efficiency that the 70 hp engine had at 70 hp. Similarly, the scaled-down engine was assumed to have the same thermal efficiency at one-half maximum power as did the 70 hp engine at half power.

Since the 70 hp engine map used in deriving fuel consumption as a function of power demand is a prechamber engine, and direct-injection (DI) engines are more fuel-efficient than indirect-injection (IDI) engines, full vehicle/engine optimization suggests

the use of DI engines. There is evidence that the fuel consumption of a DI diesel engine is approximately twelve percent lower than that of a comparable IDI diesel engine.[14]

To simulate the maximum fuel economy improvements possible, the fuel consumption rates from the 70 hp engine map were multiplied by 0.88 (reduced by 12 percent) before being entered as data for the model. In other words, it is assumed that the engines of these six optimized vehicles will be DI engines, and as such will operate with the same decrease in fuel consumption relative to IDI engines that is seen today.

After the power demand P(i), including the 0.5 hp accessory load, has been computed for a given second t=i of the driving cycle it is sent, along with the table of fuel consumption versus demanded power, to a linear interpolation subroutine. The subroutine determines the interval of power demand within the table that contains P(i) and interpolates when necessary to determine the fuel consumption rate. This rate (g/kWh) is multiplied by the power demand (kW) and divided by 3600 (s/h) to yield the fuel consumption (g) during the i^{th} second.

The resulting value for fuel consumption is returned to the main program and added to the cumulative fuel consumption. This completes the calculations for the i^{th} second, and the programs goes on to evaluation of the (i+1)th second. Fuel consumption is determined by this method for all of the time that the vehicle is in "A"-mode (accelerating or cruising). This method is also used to determine fuel consumption when the vehicle is in "B"-mode (powered deceleration), except for the case described in the following section.

Idle Fuel Flow Rate. There are several situations encountered in the course of the EPA cycles where the method described above cannot be used to determine fuel consumption. These are: (i) idling, when the vehicle velocity is zero, (ii) operation in "C"-mode, when the calculated power demand is negative, and (iii) those seconds of operation in "B"-mode where the power demand is so small (approximately 0.6 kW or less) that the interpolated value of fuel consumption is less than the idle fuel flow rate.

In these three situations, an idle fuel flow rate of 0.12 gal/hr (0.107 g/s) is assumed. The conversion of the measured 0.12 gal/hr idle fuel flow rate to 0.107 g/s is based on density of 7.078 lb/gal for diesel #2 fuel. This density is in turn based on an assumed API gravity of 35°, corresponding to the midpoint of the range of API gravity (33° to 37°) specified for diesel #2 as an EPA test fuel.

The 0.12 gal/hr idle fuel flow rate was measured in tests conducted at EPA on an Integrated Research Volkswagen (IRVW) safety vehicle.[15] This vehicle was equipped with a 70 hp turbocharged diesel engine, which is equal in maximum power to the engine of the map used to derive the table of fuel consumption as a

function of power demand. Since the maximum powers of these engines are equal, the idle fuel flow rate was adjusted in the same way that the fuel consumption values at given power demands were adjusted: The base value for idle fuel flow was multiplied by the ratio of the maximum power output of the engine sized for the vehicle to the maximum power output (70 hp) of the engine used in the determination of fuel consumption as a function of power demand.

When the vehicle is operating in "C"-mode, the scaled idle fuel consumption is added to the cumulative fuel consumption without calling the interpolation subroutine. The other two situations are treated within the subroutine: Before a one-second fuel consumption value is returned to the main program, it is checked against the scaled idle flow rate. If the calculated fuel consumption is less than that rate, the scaled idle flow rate is returned instead. This ensures that fuel consumption in any second is never less than the scaled idle fuel flow in one second.

Testing the Method. The general method was tested by using data representing a Volkswagen Rabbit. Input data was in the form of vehicle design parameters, as shown below:

mass	1077.3 kg
RRC	0.0120
$C_D A$	0.77 m^2

The rolling resistance coefficient was chosen as representative of a low (but not exceptionally low) rolling resistance tire, while the values of mass and $C_D A$ actually describe a Rabbit.

The fuel consumption versus power demand map was from a 50 hp naturally aspirated Volkswagen diesel engine.[13] This engine is very near in power to the 52 hp diesel engine that is available in MY1981 Rabbits manufactured by VWOA,[7] and hence the fuel consumption figures were not scaled. Accessory load (0.5 hp) and idle fuel flow rate (0.107 g/s) were treated in the same way as for the six optimized vehicles.

The FE projections from the energy demand model were 53 MPG for the LA4 cycle and 61 MPG for the HFET cycle. The urban FE value represents an increase of approximately 25 percent over the 42 MPG achieved by the most fuel-efficient of the MY 1981 Rabbits. [16] Since fuel economy increases of 20 percent are possible through use of CVTs,[6] the modeling approach does not appear to be overly optimistic.

Fuel economy (results). The fuel economy projections for the six optimized vehicles are shown in Table 2, along with the corresponding energy demands. The model indicates that fuel economy in the range of 90 MPG could be obtained by the two-passenger CB

Table 2

Vehicle Size Class	Assumed level of technology	Urban Cycle			HFET Cycle		
		Energy demand (MJ)	Estimated Fuel Economy (MPG) diesel	gas	Energy demand (MJ)	Estimated Fuel Economy (MPG) diesel	gas
Two-passenger	current best (CB)	2.81	92	83	4.88	87	78
	advanced (AT)	1.93	127	114	3.36	123	111
Four-passenger	CB	3.43	77	69	5.38	79	71
	AT	2.38	105	94	3.84	109	98
5 or 6-passenger and 'personal' light trucks	CB	4.21	64	57	6.48	66	59
	AT	2.88	89	80	4.46	94	85

technology vehicle; this increases to approximately 120 MPG for the advanced technology two-passenger vehicle. The increases in FE for the AT vehicles over the CB vehicles, in the other size classes, were also in the 35 to 40 percent range.

To estimate gasoline-equivalent fuel economy, the greater heat content per unit volume of diesel #2 fuel over typical gasoline was taken into account. The ratio of these heating values (138,700 BTU/gal for diesel #2, and 125,000 BTU/gal for gasoline[17]), 138.7/125 = 1.11, was multiplied by the cycle fuel consumption to estimate gasoline-equivalent fuel consumption. The resulting gasoline-equivalent fuel economy projections are also shown in Table 2.

An aspect of the optimization of all four parameters (m, RRC, C_D, A) that was not anticipated was the reversal of the ranking of urban and highway fuel economy from the commonly accepted order. For all of the vehicles in the 1981 Gas Mileage Guide[16], highway FE is greater than urban FE. This relation also holds for the two larger size classes of vehicles used in these projections, although the gap between urban and highway FE for these vehicles is relatively small. For the two-passenger vehicles, both CB and AT, projected city FE exceeds projected highway FE.

Table 3 shows the relative contributions of kinetic and aerodynamic energy demand to total energy demand for these six vehicles, over both of the EPA cycles. The kinetic energy demand dominates the urban cycle, while aerodynamic drag dominates on the highway cycle. The fuel economy reversal of the two cycles occurs because for the chosen optimized values of the vehicle parameters, the decreases in the kinetic energy requirements for urban driving are much greater than the decreases in the aerodynamic energy requirements for expressway driving.

In addition to the use of the model to estimate potential improvements in energy demand and fuel economy, there are other uses that apply to future and current vehicles, and to testing. Sensitivity of energy demand to changes in the primary vehicle parameters can be investigated, as discussed in the next section. The last section briefly touches on some of the other possible applications of the model.

B. Sensitivity Analysis

An energy demand model can be useful in investigating the impact on energy demand of changes in different vehicle parameters. The effect of small variations in aerodynamic characteristics ($C_D A$), for example, may be obscured in the "noise" of data from actual tests. This assumes that two vehicles that are identical except for the parameter of interest can be found for testing; this is often not the case, and in some instances (e.g., change frontal area only) it may be impossible.

Table 3

Vehicle Size Class	Assumed level of technology	LA4 cycle $\frac{E_K}{E}$	LA4 cycle $\frac{E_A}{E}$	HFET cycle $\frac{E_K}{E}$	HFET cycle $\frac{E_A}{E}$
Two-passenger	Current best(CB)	0.508	0.278	0.162	0.607
	Advanced(AT)	0.519	0.285	0.164	0.620
Four-passenger	CB	0.555	0.221	0.195	0.536
	AT	0.559	0.239	0.192	0.565
5/6-passenger & light truck	CB	0.565	0.210	0.203	0.521
	AT	0.578	0.218	0.206	0.538

where:

E = total energy demand over cycle (at drive wheels)
E_K = kinetic energy requirement
E_A = aerodynamic energy requirement

Table 4 presents the results of running an abbreviated sensitivity analysis using the basic energy demand model. The four-passenger vehicle with "current best technology" design parameters from the previous section was selected as the vehicle for this analysis.

First, the vehicle has its energy demand modeled using the values of mass, RRC, and C_DA as given in Table 1. The energy demands over both the LA4 and HFET cycles are then used as base values, along with the maximum power demand in each cycle. The CB four-passenger vehicle is then run through the model several additional times; in each of these runs, one of the three vehicle descriptors is either increased or decreased by ten percent. The results of the additional modelings are compared with the baseline results to examine the sensitivity of energy demand to the changes in the design parameters.

The numerical results of these modelings are listed in Table 4, along with the percent changes in energy demand and maximum power demand associated with ten percent changes in mass, RRC, and C_DA. Over the LA4 cycle, ten percent changes in vehicle mass resulted in approximately seven percent changes in energy demand. Ten percent variations in aerodynamic behavior, as characterized by C_DA, caused changes in the vicinity of three percent in energy demand; and ten percent changes in rolling resistance, as characterized by the RRC, changed energy demand by about two percent.

In the HFET cycle, with its much greater average speed and relatively minor accelerations and decelerations, changes in the aerodynamic parameter C_DA caused the greatest changes in total energy demand: approximately six percent, for ten percent variations in C_DA. Increasing or decreasing the vehicle mass by ten percent changed energy demand by just less than four percent. The percentage effects of rolling resistance changes were almost the same as for the LA4 cycle, around two percent.

C. Additional Applications

Selection of Vehicles for Testing. Another potential use of an energy-demand model is assistance in selecting candidate vehicles for some testing programs. Consider a hypothetical example: Assume that a test program has been planned to investigate the differences between dynamometer simulation accuracy for front-wheel drive (FWD) and rear-wheel drive (RWD) passenger cars; and further that sufficient test data exist to suggest the FWD vehicles experience greater loading than do RWD vehicles when tested on dynamometers. The primary objective of this hypothetical test program is to determine if FWD vehicles actually are more severely loaded on the dynamometer relative to RWDs.

By entering the available test data in this model and computing the energy demands, valuable insight could be gained as to which vehicles would best be chosen for testing. Pairs of vehicles, one FWD and one RWD, that exhibit similar internal and

Table 4

Baseline vehicle parameters:

mass	907.2	kg
RRC	0.008	
C_DA	0.68	m^2
(C_D	0.4	)
(A^D	1.7	m^2)

LA4 Cycle

Parameter	Value Rel.	Value Abs.	Total Energy Demand Abs.(MJ)	Total Energy Demand Rel.(%Δ)	Maximum Power Demand Abs.(hp)	Maximum Power Demand Rel.(%Δ)
all	baseline		3.51	0.	27.2	0.
mass	0.9	816.5	3.26	-7.1	24.7	-9.2
	1.1	997.9	3.76	+7.1	29.7	+9.2
C_DA	0.9	0.612	3.42	-2.6	27.0	-0.7
	1.1	0.748	3.61	+2.8	27.4	+0.7
RRC	0.9	0.0072	3.44	-2.0	27.1	-0.4
	1.1	0.0088	3.59	+2.3	27.4	+0.7

HFET Cycle

Parameter	Value Rel.	Value Abs.	Total Energy Demand Abs.(MJ)	Total Energy Demand Rel.(%Δ)	Maximum Power Demand Abs.(hp)	Maximum Power Demand Rel.(%Δ)
all	baseline		5.51	0.	22.6	0.
mass	0.9	816.5	5.31	-3.7	20.6	-8.8
	1.1	997.9	5.72	+3.8	24.6	+8.8
C_DA	0.9	0.612	5.20	-5.7	22.4	-0.9
	1.1	0.748	5.85	+6.2	22.8	+0.9
RRC	0.9	0.0072	5.39	-2.2	22.4	-0.9
	1.1	0.0088	5.63	+2.1	22.7	+0.4

external characteristics but have modeled energy demands that differ greatly would be good test candidates, as would any FWD vehicle whose modeled energy demands differed widely depending on whether road-derived or dynamometer-derived data were input. By this exercise, the vehicles that are most likely to demonstrate the suggested problem will be selected for testing. (This is cited only as an example. The FWD/RWD simulation accuracy is dependent on the distribution of vehicle weight between the front and rear axles.)

In a similar vein, this model could aid in the discretionary selection of vehicles for confirmatory testing. For all test vehicles, the required information for modeling energy demand on the dynamometer is available. Vehicles that display ratios of modeled energy demand to fuel economy (MJ/MPG) that are unusually high or low would be obvious candidates for confirmatory testing.

The question that can be answered using the model is: Based on the modeling parameters, which of the vehicles of the possible test group are most likely to exhibit the highest levels of NOx emissions and the lowest fuel economy? Consider a pair of vehicles sharing engine and transmission/drivetrain characteristics in common, one of which will be a discretionary choice to be tested. In such a case, vehicle A may have a greater mass than vehicle B, while vehicle B has a higher value for total dynamometer power absorption. By modeling the energy demand of these vehicles and choosing the vehicle with greater energy demand for confirmatory testing, the lower of the two in fuel economy (and probably the greater in NOx emissions) will be tested.

Test result guidelines. When a fuel economy-related test program is planned, the test vehicles could be run through the model before testing commenced. The results of the pre-test modeling can serve as guidelines in evaluation of the test data, flagging test results that deviate beyond reasonable allowances from the modeled output. Such a pre-test exercise might allow earlier detection and correction of errors that otherwise may have gone undetected, and been suspected only at the conclusion of the program. Another benefit is that more test time can then be concentrated in areas, or on vehicles, where apparently anomalous results are occurring.

This is not a complete list of the potential uses of a model of this type. In general, it is attractive because it provides insight into many aspects of vehicle energy demand at low cost.

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```
      C        ******   LDV ENERGY DEMAND MODEL   ******
      C
      C        VARIABLE DIMENSIONS:  VMASS, IMASS(KG); CDA(M**2); RHO(KG/M**3);
      C          D1(M); D2(MI); V, DVDT, VMEAN(M/S); A0(MI/HR-S); A2(HR/MI-S);
      C          F0, FTOT, FCON, FVSQ, FORCE, FORCE2, TIREF, AEROF, FINER(N);
      C          F2(KG/M); POWER, POWER2, TIREP, AEROP, PINER, AHPW(W); TIME,
      C          TMAX, A, B, C(S); KW, SKW(KW); ENERGY, TIRENG, AEROE, POSKIN,
      C          NETKIN(J); ACC, POWHP, PMAX, AHP(HP); FUEL, SFUEL(G); IDLE,
      C          SIDLE(G/S); FC(G/KW-HR); MPG(MI/GAL).
      C
      C        I/O ASSIGNMENTS:  6 = OUTPUT FILE
      C                          5 = INPUT FILE: DRIVING CYCLE
      C                          4 = INPUT FILE: VEHICLE DATA
      C                          3 = INPUT FILE: NCAR, NSEC, DTYP
      C                          2 = INPUT FILE: FC/PWR TABLE
            REAL V(2000)
            INTEGER A, B, C, I, J, K, L
            INTEGER NCAR, NSEC, DTYP, VEHID, TMAX
            REAL VMASS, IMASS, PMAX
            REAL D1, D2, DVDT, VMEAN
            REAL TIREF, TIREP, TIRENG
            REAL AEROF, AEROP, AEROE
            REAL FORCE, POWER, ENERGY
            REAL FORCE2, POWER2, POWHP
            REAL*8 FINER, PINER, POSKIN, NETKIN
            REAL F2, RRC, CDA, RHO
            REAL AHP, AHPW, TIME
            REAL A0, A2, FCON, FVSQ, FTOT
            INTEGER NMAP, FLAG
            REAL KW(15), FC(15), MPG, FUEL, SFUEL, IDLE
            REAL SKW(15), SCALE, SIDLE, BTU, MAXHP, MAPHP
      C
      C        A,B,C USED TO DETERMINE TIME SPENT IN EACH MODE:
      C         A-MODE:  FORCE > 0 & FINER > 0
      C         B-MODE:  FORCE > 0 & FINER < 0
      C         C-MODE:  FORCE < 0 & FINER < 0
      C
            A = 0
            B = 0
            C = 0
      C
      C        READ IN KW/FC TABLE FOR INTERP
      C
            READ(2,100) MAPHP, NMAP
            READ(2,103) (KW(I),FC(I), I=1,NMAP)
            IDLE = 0.107
            BTU = 1.11
            RHO = 1.17
      C
      C        NCAR = NUMBER OF CASES IN RUN
      C        NSEC = LENGTH OF DRIVING CYCLE
      C        DTYP = 1 (DYNAMOMETER COASTDOWN DATA),
      C               2 (ROAD COASTDOWN DATA), OR
      C               3 (VEHICLE DESIGN PARAMETERS)
      C
            READ(3,106) NCAR, NSEC, DTYP
      C
      C        READ DRIVING CYCLE, CONVERT MI/HR TO M/S
      C
            READ(5,109) (V(I), I=1,NSEC)
```

```
      DO 1  I = 1, NSEC
         V(I) = 0.447*V(I)
    1 CONTINUE
C
C       LOOP:  ONCE PER VEHICLE
C
      DO 10  K = 1, NCAR
C
      WRITE(6,112) K
C
C        TYPE OF INPUT DATA
C
      IF (DTYP.EQ.3)  GO TO 3
      IF (DTYP.EQ.2)  GO TO 2
C
C        DYNAMOMETER COASTDOWN DATA
C
      READ(4,115) VEHID, MAXHP, VMASS, AHP, TIME
      FTOT = (VMASS*16093.4)/(TIME*3600.)
      AHPW = AHP*745.7
      FVSQ = AHPW/22.35
      FCON = FTOT-FVSQ
      F0 = FCON
      F2 = FVSQ/499.5
      RRC = F0/(VMASS*9.81)
      CDA = (F2*2.)/RHO
      IMASS = 1.018*VMASS
      WRITE(6,118) VEHID, VMASS, MAXHP
      WRITE(6,121) F0, F2
      WRITE(6,124) AHP, TIME
      WRITE(6,127) RRC, CDA
         GO TO 4
C
C        ROAD COASTDOWN DATA
C
    2    CONTINUE
      READ(4,130) VEHID, MAXHP, VMASS, A0, A2
      F2 = VMASS*A2*(3600./1609.34)
      F0 = VMASS*A0*(1609.34/3600.)
      RRC = F0/(VMASS*9.81)
      CDA = (F2*2.)/RHO
      IMASS = 1.035*VMASS
      WRITE(6,118) VEHID, VMASS, MAXHP
      WRITE(6,121) F0, F2
      WRITE(6,133) A0, A2
      WRITE(6,127) RRC, CDA
         GO TO 4
C
C        VEHICLE DESIGN PARAMETERS
C
    3    CONTINUE
      READ(4,136) VEHID, MAXHP, VMASS, RRC, CDA
      F0 = VMASS*9.81*RRC
      F2 = 0.5*RHO*CDA
      IMASS = 1.035*VMASS
      WRITE(6,118) VEHID, VMASS, MAXHP
      WRITE(6,121) F0, F2
      WRITE(6,139) RRC, CDA
C
    4    CONTINUE
```

```
      C
      C       SCALE FC/KW ARRAY, IDLE FC
      C
            SCALE = MAXHP/MAPHP
            SIDLE = IDLE*SCALE
            DO 5  L = 1, NMAP
               SKW(L) = KW(L)*SCALE
          5 CONTINUE
            WRITE(6,142) SCALE
      C
      C       INITIALIZE RUNNING SUMS, PMAX, TMAX TO ZERO
      C
            D1 = 0.
            AEROE = 0.
            NETKIN = 0.
            POSKIN = 0.
            TIRENG = 0.
            ENERGY = 0.
            ENGINE = 0.
            SFUEL = 0.
            PMAX = 0.
            TMAX = 0
      C
      C       LOOP:  ONCE PER SEC OF CYCLE
      C
            DO 9 I = 1, NSEC
      C
      C          CALCULATE DVDT, VMEAN, & DISTANCE TRAVELLED
      C
               IF (I.EQ.1)  GO TO 11
               J = I-1
               DVDT = V(I)-V(J)
               VMEAN = (V(I)+V(J))/2.
               GO TO 22
           11  CONTINUE
               DVDT = V(I)
               VMEAN = V(I)
           22  CONTINUE
               D1 = D1+VMEAN
      C
      C          CALCULATE FORCES AT ITH SECOND
      C
               TIREF = F0
               AEROF = F2*(VMEAN**2)
               FINER = IMASS*DVDT
               FORCE = TIREF+AEROF+FINER
      C
      C          CHECK SIGNS OF FORCE AND FINER
      C
               IF (FORCE.LE.0.)  GO TO 7
               IF (FINER.LE.0.)  GO TO 6
      C
      C          FORCE AND FINER ARE POSITIVE: CALCULATE
      C          ALL COMPONENTS OF FORCE AND ENERGY
      C
               A = A+1
               POWER = FORCE*VMEAN
               POWER2 = POWER/0.9
               POWHP = POWER2/745.7
      C
```

```
                POWHP = POWHP+0.5
C
                IF (POWHP.GT.PMAX)  TMAX = I
                IF (POWHP.GT.PMAX)  PMAX = POWHP
                POWER2 = POWHP*745.7
                CALL INTERP(SKW,FC,POWER2,SIDLE,NMAP,FUEL,FLAG)
                SFUEL = SFUEL+FUEL
                IF (FLAG.EQ.1)  WRITE(6,199) I
                TIREP = TIREF*VMEAN
                PINER = FINER*VMEAN
                AEROP = AEROF*VMEAN
                TIRENG = TIRENG+TIREP
                NETKIN = NETKIN+PINER
                POSKIN = POSKIN+PINER
                AEROE = AEROE+AEROP
                ENERGY = ENERGY+POWER
                GO TO 8
C
C                 FORCE IS POSITIVE AND FINER IS NEGATIVE:  PROPORTION
C                 ENERGY DEMAND TO ROLLING AND AERODYNAMIC RESISTANCES
C
          6     CONTINUE
                B = B+1
                FORCE2 = TIREF+AEROF
                TIREF = TIREF*(FORCE/FORCE2)
                AEROF = AEROF*(FORCE/FORCE2)
                TIREP = TIREF*VMEAN
                PINER = FINER*VMEAN
                AEROP = AEROF*VMEAN
                POWER = FORCE2*VMEAN
                POWER2 = POWER/0.9
                POWHP = POWER2/745.7
C
C                 ADD ACCESSORY LOAD TO POWER DEMAND
                  POWHP = POWHP+0.5
C
                IF (POWHP.GT.PMAX)  TMAX = I
                IF (POWHP.GT.PMAX)  PMAX = POWHP
                POWER2 = POWHP*745.7
                CALL INTERP(SKW,FC,POWER2,SIDLE,NMAP,FUEL,FLAG)
                SFUEL = SFUEL+FUEL
                IF (FLAG.EQ.1)  WRITE(6,199) I
                TIRENG = TIRENG+TIREP
                NETKIN = NETKIN+PINER
                AEROE = AEROE+AEROP
                ENERGY = ENERGY+POWER
                GO TO 8
C
C                 FORCE IS NEGATIVE:  CALCULATE INERTIA TERMS AS CHECK
C
          7     CONTINUE
                C = C+1
                PINER = FINER*VMEAN
                NETKIN = NETKIN+PINER
                SFUEL = SFUEL+SIDLE
C
C                 (THREE BRANCHES REJOINED)
C
          8     CONTINUE
```

```
      9 CONTINUE
C
C       OUTPUT:  BREAKDOWN OF ENERGY DEMANDED
C
        WRITE(6,145) ENERGY,TIRENG,AEROE,NETKIN,POSKIN
C
C       COMPENSATE FOR ESTIMATED CVT EFFICIENCY
C
        ENERGY = ENERGY/0.9
        WRITE(6,148) ENERGY
        D2 = D1/1609.34
        WRITE(6,151) D1, D2
        WRITE(6,154) A, B, C
        WRITE(6,157) PMAX, TMAX
C
C       CONVERT FC TO FE
C
        SFUEL = SFUEL/3218.
        MPG = D2/SFUEL
        WRITE(6,160) SFUEL, MPG
        SFUEL = SFUEL*BTU
        MPG = D2/SFUEL
        WRITE(6,163) SFUEL, MPG
C
C       RESET "MODE" INDICES
C
        A = 0
        B = 0
        C = 0
C
C       END OA LOOP
C
     10 CONTINUE
    100 FORMAT(F5.1,I6)
    103 FORMAT(F5.1,F6.1)
    106 FORMAT(3I5)
    109 FORMAT(16F5.1)
    112 FORMAT('CASE NUMBER',I4)
    115 FORMAT(I5,F5.1,F10.1,F5.1,F10.2)
    118 FORMAT('VEHICLE #',I5,5X,'MASS:',F7.1,' KG',5X,'MAX HP:',F5.1,' HP'
    121 FORMAT(3X,'F0 =',F7.1,' N',5X,'F2 =',F7.4,' KG/M',10X,'(CALC)')
    124 FORMAT(3X,'AHP =',F5.1,' HP',4X,'55-45 CD TIME:',F6.2,' S',4X,'(DATA
    127 FORMAT(3X,'RRC =',F8.5,4X,'CDA =',F6.3,' M**2',10X,'(CALC)')
    130 FORMAT(I5,F5.1,F10.1,F10.4,E12.4)
    133 FORMAT(3X,'A0 =',F7.4,' MI/HR-S',3X,'A2 =',E12.4,' HR/MI-S',5X,'(DAT
    136 FORMAT(I5,F5.1,F10.1,F10.5,F10.3)
    139 FORMAT(3X,'RRC =',F8.5,4X,'CDA =',F6.3,' M**2',11X,'(DATA)')
    142 FORMAT(3X,'(SCALING FACTOR',F8.4,')')
    145 FORMAT('ENERGY DEMANDS:  TOTAL:',E12.4,3X,'TIRE:',E12.4,3X,'AERO:',E
       */39X,'NETK:',E12.4,3X,'POSK:',E12.4)
    148 FORMAT(16X,'(TOTAL:',E12.4,3X,'AT ENGINE, W/ 90% EFF CVT)')
    151 FORMAT(17X,'DISTANCE:',8X,F7.1,' M',6X,F5.2,' MI')
    154 FORMAT(17X,'TIME IN MODE:',4X,'A:',I5,' S',6X,'B:',I5,' S',6X,'C:',I
    157 FORMAT(17X,'MAX POWER DEMAND:',F7.1,' HP AT T =',I5,' S')
    160 FORMAT(17X,'(DIESEL #2) EST FC:',F9.4,' GAL, EST FE:',F8.1,' MPG')
    163 FORMAT(17X,'(GAS EQUIV) EST FC:',F9.4,' GAL, EST FE:',F8.1,' MPG')
    199 FORMAT(3X,'*** WARNING: POSSIBLE ERROR IN FC/FE AT T =',I5,' S ***')
        RETURN
        END
C
```

```
C       FIND FUEL CONSUMPTION USING TABLE FROM ENGINE MAP
C
      SUBROUTINE INTERP(IKW,IFC,IP,IFLOW,IMAP,IFUEL,IFLAG)
      REAL IKW(15), IFC(15), IP, IFLOW, IFUEL
      REAL X1, X2, DX, Y1, Y2, DY, M, B
      INTEGER G, N, IMAP, IFLAG
      IFLAG = 0
C
C       CONVERT POWER FROM W TO KW, FIND CORRECT INTERVAL
C        (WHEN VELOCITY IS ZERO USE IDLE FUEL FLOW RATE)
C
      IF (IP.LE.0.)  FUEL = IFLOW
      IF (IP.LE.0.)  GO TO 204
      IP = IP/1000.
      N = IMAP-1
      DO 201  G = 1, N
         IF ((IP.GT.IKW(G)).AND.(IP.LT.IKW(G+1)))  GO TO 202
  201 CONTINUE
C
C       INTERVAL OF IP NOT FOUND: ERROR MESSAGE VIA FLAG
C
      IFLAG = 1
      GO TO 203
C
C       COMPUTE EQN OF INTERPOLATING LINE
C
  202 CONTINUE
      Y1 = IFC(G)
      Y2 = IFC(G+1)
      X1 = IKW(G)
      X2 = IKW(G+1)
      DY = Y2 - Y1
      DX = X2 - X1
      M = DY/DX
      B = Y1 - (M*X1)
C
C       IFUEL IS RETURNED WITH FUEL CONS AT ITH SEC
C
      IFUEL = B + (M*IP)
C
C       CONVERT IFUEL FROM G/KW-HR TO G/S
C       (FC MUST BE > OR = IDLE FC RATE)
C
      IFUEL = (IFUEL*IP)/3600.
      IF (IFUEL .LT. IFLOW)  IFUEL = IFLOW
      GO TO 204
  203 CONTINUE
      IFUEL = (IFC(IMAP)*IP)/3600.
  204 CONTINUE
      RETURN
      END
```